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Reduction of System Inherent Pressure Losses at Pressure Compensators of Hydraulic Load Sensing Systems

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Abstract

In spite of their high technical maturity, load sensing systems (LS) have system-inherent energy losses that are largely due to the operation of parallel actuators with different loads at the same pressure level. Hereby, the pressure compensators of the system are crucial. So far, excessive hydraulic energy has been throttled at these compensators and been discharged as heat via the oil. The research project “Reduction of System Inherent Pressure Losses at Pressure Compensators of Hydraulic Load Sensing Systems” aims to investigate a novel solution of reducing such energy losses. The pressure of particular sections can be increased by means of a novel hydraulic circuit. Therefore, a recovery unit is connected in series with a hydraulic accumulator via a special valve in the reflux of the actuators. The artificially increased pressure level of the section reduces the amount of hydraulic power to be throttled at the pressure compensators. As long as a section fulfills the switching condition of the valve, pressure losses at the respective pressure compensator can be reduced. Thus, via a suitable recovery unit excessive energy can be regenerated and can be directed to other process steps eventually.

KEYWORDS: Efficiency Improvement, Load Sensing, Pressure Compensators, Energy Losses, Energy Recuperation

1. Motivation and Basic Principles

At low energy losses, load sensing (LS) systems enable the simultaneous use of actuators connected in parallel. The system pressure always adjusts to the pressure of the actuator with the highest load level plus the LS pressure difference Δp_{LS}. All further actuators are fed at system pressure level as well. In each section, hence, the system pressure must be throttled down to the respective sectional pressure /1/.

The pressure compensators (PC) of each section, serving as hydraulic resistors in the respective flow paths, are the central components. The main functions of (upstream) pressure compensators are to adjust system and section pressures and to establish a constant pressure difference at the section valves. Especially when system pressure is much higher than section pressure, considerable energy losses may be caused. This typically occurs whenever differing loads are operated simultaneously /2/.

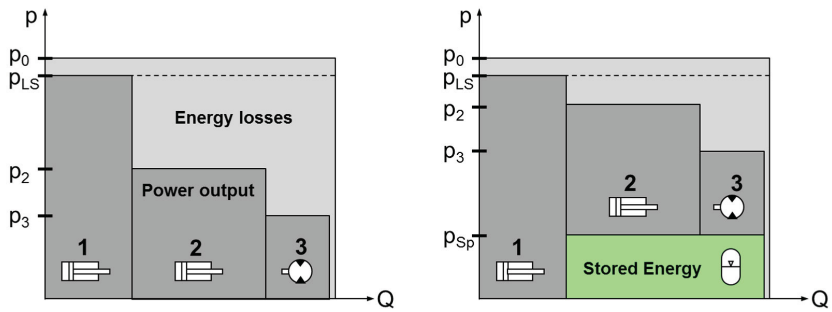


Figure 1: Power loss of a conventional LS system without (left) and with (right) accumulator series-connected to two actuators.

By way of example, **Figure 1** shows the power requirement of three actuators. The left part of the figure shows the situation for a conventional LS system. In the magnitude of the LS pressure difference, the system pressure level p_0 is above the pressure level of the actuator with the highest load p_{LS}. Actuators 2 and 3 require the lower pressure levels p_2 and p_3. The right part of the figure shows these actuators to be connected in series with an accumulator of the pressure p_{Sp}, which leads to increased section pressures causing reduced pressure differences at the pressure compensators and thus reduced losses.

Within the research project “Reduction of System Inherent Pressure Losses at Pressure Compensators of Hydraulic Load Sensing Systems” that is funded by the VDMA Fluid Technology Research Fund, such a circuit is being developed /3/. The target applications feature single-circuit LS systems with multiple simultaneously operating hydraulic actuators, which are state of the art in all kinds of compact machines, mid-range hydraulic excavators, agricultural and forestry equipment or municipal vehicles.

During the project, the basic circuit parameters are determined and preliminary examinations are conducted by means of simulations. Based on the results, a prototype of the developed circuit is being set up on a test rig and being tested. For details, see /4/.

2. Mode of Operation and Implementation of Circuit

Figure 2 shows the LS system of a mobile machine with upstream pressure compensators in the investigated efficiency-improved configuration.

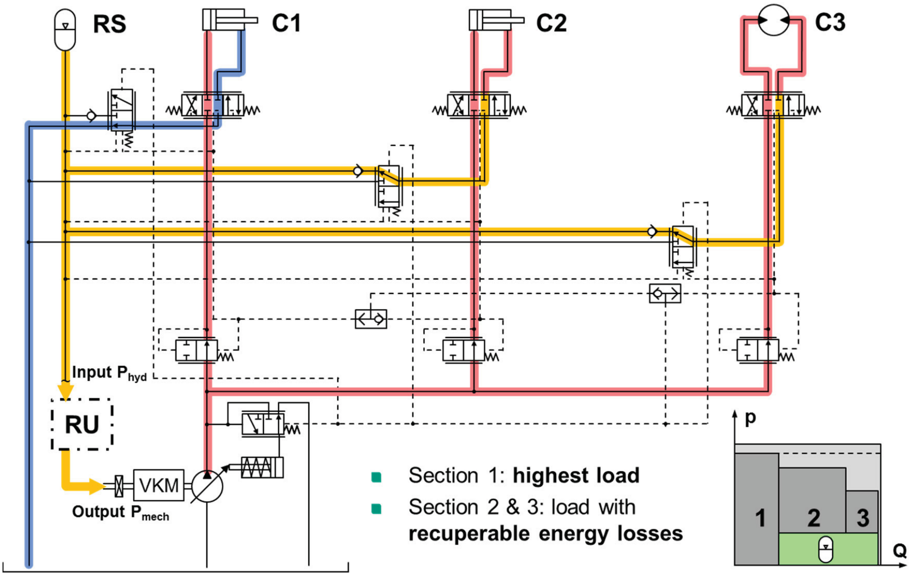


Figure 2: LS system with increased efficiency.

The diagram depicts the actuator sections (C1 – C3) and the hydraulic LS pressure supply. Whereas C1 is the actuator with the highest load, C2 and C3 currently operate at low loads and are thus connected to the recovery section (RS). The left part of the picture shows the recovery section of the system, consisting of an accumulator and the recovery unit (RU). The development of the recovery unit is not part of the current project. However, a variable displacement motor, which is connected to the combustion engine via a (switchable) clutch could be used for recuperation (see also /5/).

3. Circuit Design

Figure 3 shows the design of the circuit of a particular actuator section in detail. The central component is the tank/accumulator logic valve (T/A-LV) which is integrated in the reflux pipe of each section. The function of the valve is to differentiate between actuators of low and high load pressure. Depending on the load level, the valve switches the reflux of the actuator either to the tank or to the recovery unit. Thereby a sufficient pressure loss at the pressure compensators must be guaranteed for correct functionality of the LS principal.

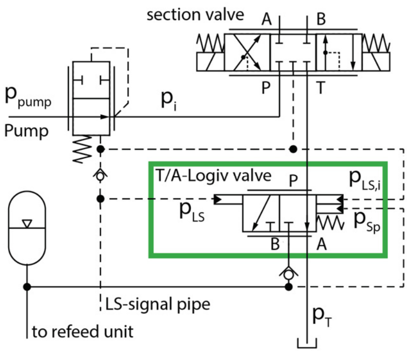


Figure 3: Design of the circuit (conservative principle).

The valve works according to the following switching condition:

$$\boldsymbol{p_{LS} * A > p_{Sp} * A * x_A + p_{LS,i} * A + F_{Spring}} \quad with\ x_A = \frac{A_{Piston}}{A_{Ring}}, \tag{1}$$

p_{LS} is the LS system pressure, p_{Sp} is the pressure in the recovery section or accumulator, $p_{LS,i}$ is the section pressure of a actuator i, x_A is the surface ratio of a differential cylinder, and F_{Spring} is a spring force which additionally acts on the spool of the valve. From (1) three load cases can be derived for the individual actuators:

$$Case\ 1: \quad \boldsymbol{p_{LS} < p_{Sp} * x_A + p_{LS,i} + \frac{F_{Spring}}{A}} \tag{2}$$

When an actuator fulfills this condition, lost energy cannot be recovered (i.e. at the actuator with the highest load). To ensure that the overall energy consumption of the system does not increase, the reflux of the actuator with the highest load is generally directed to the tank completely. However, the T/A-LV throttles the reflux of all other actuators, which fulfill the above condition but do not define the LS pressure level.

$$Case\ 2: \quad \boldsymbol{p_{LS} = p_{Sp} * x_A + p_{LS,i} + \frac{F_{Spring}}{A}} \tag{3}$$

Case 2 defines the threshold switching condition. Theoretically, this case represents the situation in which the metering edge of the tank is fully closed, while the metering edge of the accumulator is fully opened.

$$Case\ 3: \quad \boldsymbol{p_{LS} > p_{Sp} * x_A + p_{LS,i} + \frac{F_{Spring}}{A}} \tag{4}$$

When an actuator of lower load satisfies this condition, lost energy can be recuperated. The T/A-LV tank edge is closed so that the reflux of the actuator is fed into the recovery unit. The pressure in the recovery section now directly counteracts the supply pressure of the connected actuator(s), thus increasing the section pressure.

Within the present project, the increase of the section pressure level is simulated by means of a proportional pressure-limiting valve. Suitable methods for refeeding the recovered energy will be investigated in a following project.

Two different circuit principles, the **conservative** and the **adaptive** one (**Figure 4**), are being examined, which differ in the way of taking the pressure translation properties of differential cylinders into account. However, this has no influence on rotatory actuators or double-rod cylinders.

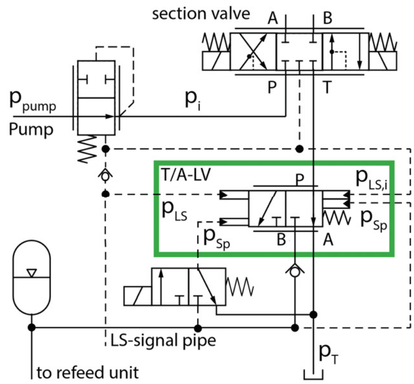


Figure 4: Design of the circuit (adaptive principle).

The pressure of the recovery section is applied to the piston surface corresponding to the reflux, i.e. to the piston side during instroke and to the ring side during outstroke. Hence, the pressure increase in the cylinder inlet also depends on the direction of motion. For example, in case of a cylinder with a surface ratio of $x_A = 2$, the pressure in the reflux line is doubled during instroke and reduced by half during outstroke (**Figure** 5).

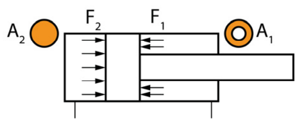


Figure 5: Differential cylinder as pressure translator.

Since the inlet pressure equals the LS pressure $p_{LS,i}$, the direction of motion affects the pressure difference which is necessary at the pressure compensator to establish a connection of the actuator with the recovery unit. Compared to the conservative principle, the T/A-LV of the adaptive principle has one additional hydraulic control surface, which is connected either to the pressure level of the recovery section or to the tank. The connection is established by an additional switching valve in dependence on the direction of motion. Thus, during outstroke, when the additional control surface is connected to the recovery unit, the recovery pressure induces different forces on the opposite sides of the valve spool. By using the left control surface with a size of $(x_A - \frac{1}{x_A}) * A$, where A equals the size of the right surface (ref. to Figure 6), the sum of both forces equals a force applied on the left side with an effective surface of $\frac{1}{x_A} * A$.

According to (1), p_{Sp} is part of the switching condition. Furthermore, p_{Sp} has a direct influence on $p_{LS,i}$. For a differential cylinder with passive load, $p_{LS,i}$ can be defined ideal as:

$$p_{LS,i} = p_{Load} + p_{Sp} * {x_A}^s; \quad with\ s = \begin{cases} -1\ during\ Outstroke \\ +1\ during\ Instroke \end{cases} \tag{5}$$

For the conservative principle, equation (1) can be transformed with (5) to:

$$p_{LS} * A > p_{Load} * A + p_{Sp} * A * (x_A + {x_A}^s) + F_{Spring} \tag{6}$$

For the adaptive principal, equation (1) can be transformed with (5) to:

$$p_{LS} * A > p_{Load} * A + p_{Sp} * A * 2 * {x_A}^s + F_{Spring} \tag{7}$$

Both equations (6), (7) differ in the terms concerning ${x_A}^s$ and p_{Sp}:

$$(x_A + {x_A}^s) \geq 2 * {x_A}^s, \quad for\ all\ x_A \geq 1, \quad s = \begin{cases} -1\ during\ Outstroke \\ +1\ during\ Instroke \end{cases} \tag{8}$$

Therefore, the switching condition of the adaptive principle allows for a higher external load during outstroke than the switching condition of the conservative principle. Thus, the adaptive principle has a higher efficiency improvement potential during outstroke (compare /4/). However, the complexity of the adaptive T/S-LV and of its circuit is higher.

4. Losses in Different Operating Cycles

Within the research project, different mobile machines are investigated with regard to the applicability of the circuit by analyzing both the system architecture and the typical operating cycles. Application of the circuit is conceivable and practicable in machines with one-circuit LS-system (of any configuration) and in machines with multiple simultaneously operated actuators at different load levels in one LS circuit. For the current research project, a hydraulic excavator is used as reference system. Typical operating cycles were taken from /6/. **Table 1** shows the results of cycle loss analyses.

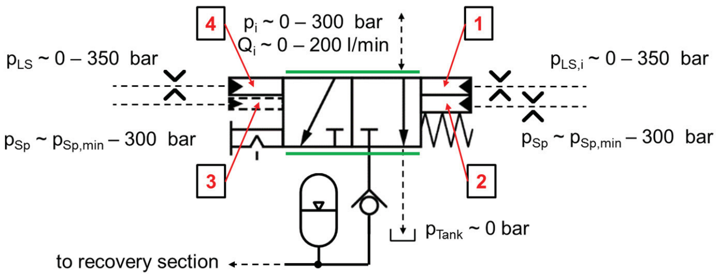
Cycle	90°-work cycle	Plane
Total energy consumption	1480 kJ	227 kJ
System-inherent losses:	294 kJ	27 kJ
• Arm section	57 kJ	5 kJ
• Boom section	5 kJ	22 kJ
• Bucket section	176 kJ	not specified
• Swing drive section	54 kJ	not specified

Table 1: Energy loss analysis of Cycles from /6/, operator A

The investigation has shown that approximately 10% – 20% of the input energy is lost at the pressure compensators of the machine. Further losses in the system are e.g. due to leakage and are therefore not considered.

5. Tank/Accumulator Logic Valve

The tank/accumulator logic valve (T/A-LV) is a central component of the novel circuit. For easy integration of the circuit into existing systems, the valve comprises hydraulic actuation. The integrated design must be compact to fit the typically very limited installation space of mobile machines. In addition, the valve must have vibration-damping properties. The flow parameters are shown in **Figure 6**.



Principle	Control Surface 1	2	3	4
Conservative	A	A * x_A		A
Adaptive	A	A * x_A	A * (x_A – 1/x_A)	A

Figure 6: T/A-LV, adaptive principle (with control surface no. 3).

Depending on the respective design (conservative or adaptive circuit principle), the valve requires three or four hydraulic control surfaces. Switching valves could principally be used in the circuit, but so far no satisfying parameter setting could be found. Because of that, the switching operation causes vibrations in the entire system, which lead to a reduced performance. Therefore, the T/A-LV is designed as a proportional valve in the present project. Further damping can be achieved by integrating orifices into the control ports.

Within this project, the conceptional design of the T/A-LV has been developed. The left side of **Figure 7** shows the valve characteristics determined for the metering edges P/T (red, reflux to tank) and P/A (blue, reflux to recovery unit). The valve-characteristics can be obtained by axially displaced triangular notches in the spool (ref. to Figure 7, right).

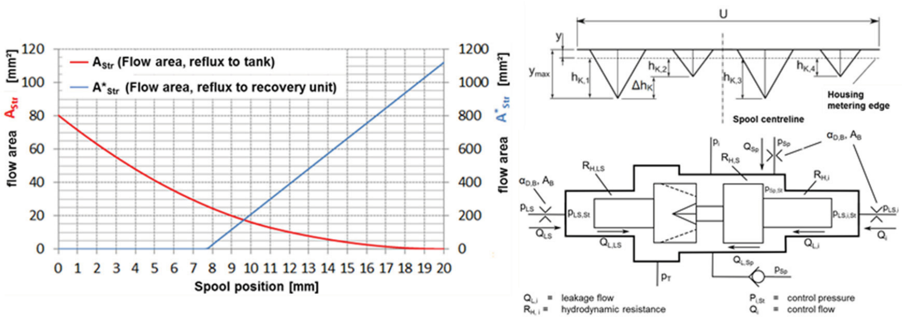


Figure 7: T/A-LV spool geometry.

Due to the high complexity of the design, in the present project the T/A-LV is represented by means of separate metering edges (ref. to Figure 11) instead of an integrated valve. The hardware valve will be developed further in a following project.

6. Simulation Results

Characteristic results of the simulations carried out within this project are presented below. These results were determined by means of two different models or degrees of model details, respectively:

- Model 1: Simplified test rig simulation model
- Model 2: Hydraulic excavator simulation model with multi-body model in co-simulation

The software DSHplus 3.9 by Fluidon was used for hydraulic simulations. For co-simulation, the programs Matlab / Simulink by Mathworks as well as the multi-body simulation program SIMPACK 9.5.1 by SIMPACK were used in addition.

6.1. Model 1: Comparison Between Conventional and Optimized LS Systems

In the following figure, the optimized (adaptive) system is compared to a conventional LS system. A ramp-type external load acting on the actuator has been selected such that the actuator, a cylinder during outstroke, allows for recuperation at the beginning but exceeds the recuperation limit towards the end of the load cycle. The first diagram of **Figure 8** shows the position x and the velocity v of the piston rod during outstroke for the conventional (dashed) and the optimized LS-System. Figure 8, middle shows the pressure in the cylinder and the corresponding external load. In Figure 8 bottom, the position of the T/A-LV spool can be seen. In all diagrams, the x-axis shows the simulation time during the load cycle.

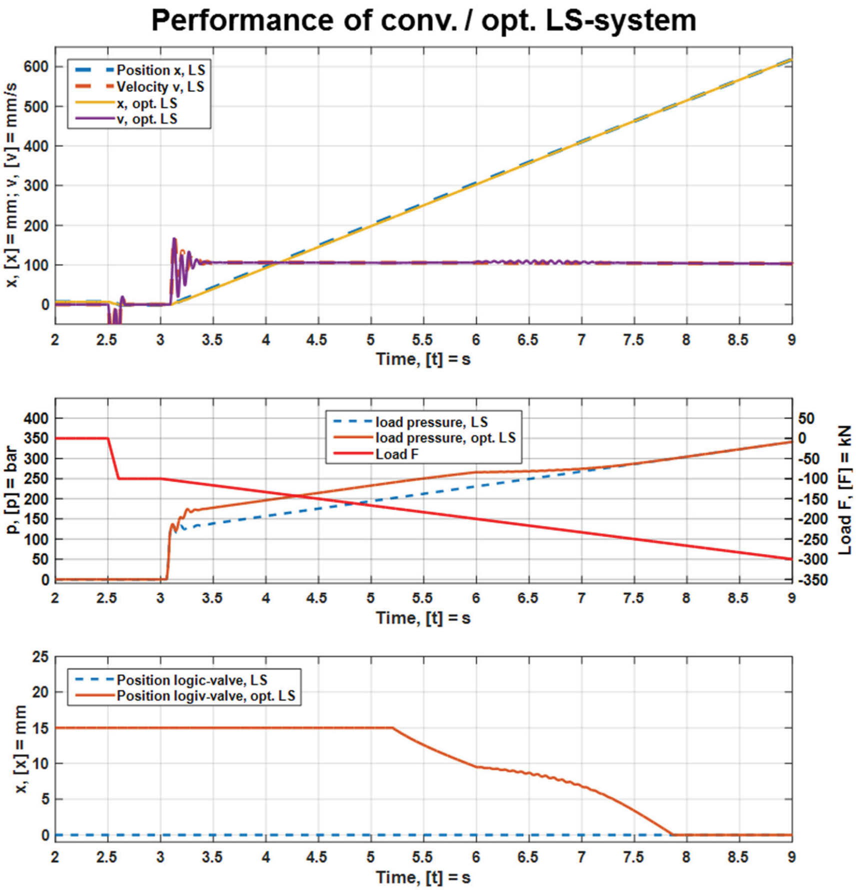


Figure 8: Circuit effect on actuator movements.

The rod movements of the conventional and optimized LS are almost identical (ref. to Figure 8, top). In the range of 3 s to approximately 7 s, the section pressure of the actuator is increased by the amount of the pressure in the recovery section (about 50 bar, ref. to Figure 8, middle). At about 5.3 s and above, the external load reaches the threshold of the switching condition i.e., the T/A-LV closes the accumulator pipe because the pressure of the actuator approaches the LS pressure. In the range of about 6 s to 7 s, vibrations at the T/A-LV occur (Figure 8, bottom) because damping of the valve has not yet been optimally adjusted. Nevertheless, the vibrations in the reflux circuit do not have any significant effect on the actuator considered or on other actuators in the investigated system.

6.2. Model 1: Efficiency Improvement in an LS System

The efficiency improvement potential was investigated by means of another synthetic load case. **Table 2** shows the results obtained.

	conventional LS	optimized LS
Actuator 1: highest load	p = 320 bar, Q = 38 l/min	
Actuator 2: constant force,	Outstroke: p = 124 bar, Q = 103 l/min	
F = 100 kN	Instroke: p = 60 bar ; Q = 74 l/min	
Total of consumed energy, system	875 kJ	877 kJ
Total of recuperated energy	0 kJ	**91 kJ**
Possible efficiency improvement	0 %	**10 %**

Table 2: Efficiency improvement simulation.

By means of the circuit, the energy consumption of the system could be reduced by 10 %, depending on the efficiency rate of all components of the recovery section.

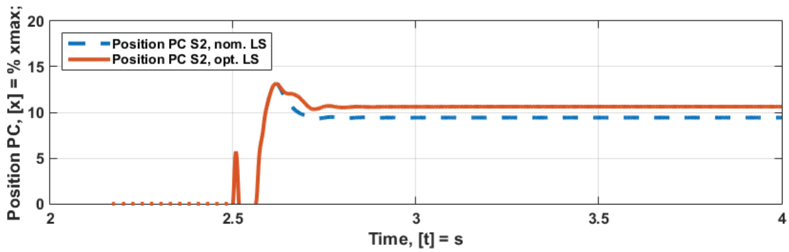


Figure 9: Influence of the circuit on a sectional pressure compensator.

Figure 9 depicts the influence of the pressure increase on the deflection of the pressure compensator of section 2. It is evident that in the optimized case, the pressure compensator is more widely open so that lower pressure losses occur.

6.3. Model 2: Performance of Optimized System (Adaptive) in Co-simulation

To evaluate the performance of the optimized system, a hydraulic model was combined with the multi-body model of a hydraulic excavator. Suitable cycle data was taken from /6/. **Figure 10** shows selected simulation results.

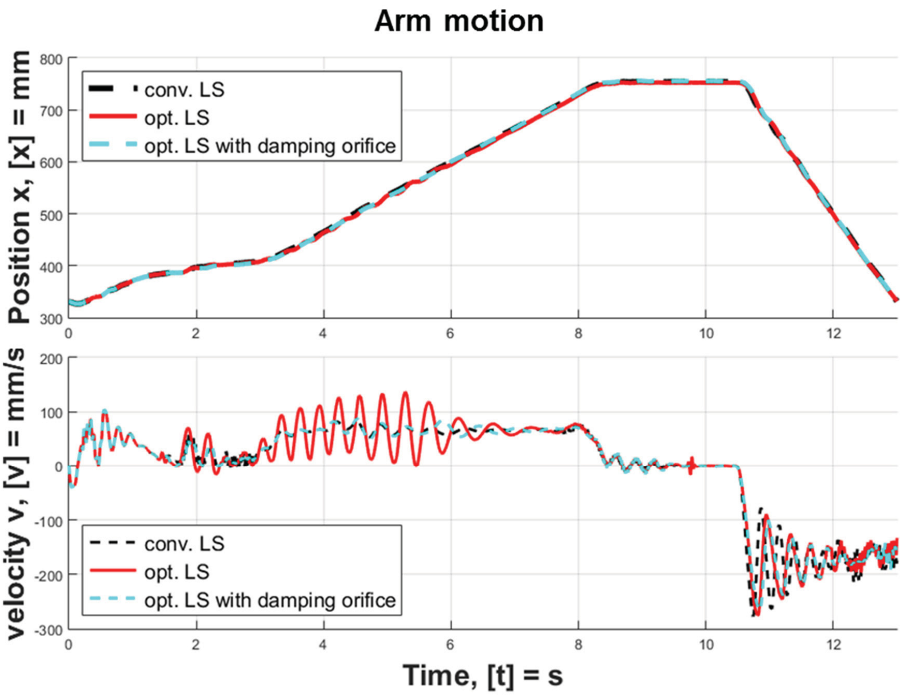


Figure 10: Comparison of arm cylinder movements by motion and velocity.

The first diagram shows the motion of the arm cylinder in the conventional LS as well as in the optimized LS with and without dampening. The second diagram shows the velocity of the arm cylinder in the same way. Simulation results show that vibrations are strongest in the arm section. This is evident from the deviations of the cylinder movement (red line) from the nominal curve (black line) in the period from about 4 s to 7 s. With adequate T/A-LV adjustment and especially when using additional damping orifices in the control ports of the valve, vibrations were considerably reduced so that the performance level of both systems (conventional and optimized) were almost identical, which can be seen in the reduced velocity vibrations of the cylinder.

7. T/A-LV Prototype

Figure 11 shows the draft of the T/A-LV functional prototype for the test rig. Since development and manufacturing of a spool valve with three or four control surfaces is quite complex and costly, the functional prototype is provided with separate metering edges by using two electrohydraulic proportional throttle valves. In addition, a pressure control valve is integrated for maximum pressure control in case of malfunctions. The different pressure levels in the system are being monitored by pressure transducers, while the evaluation of the switching condition is achieved by the valve control.

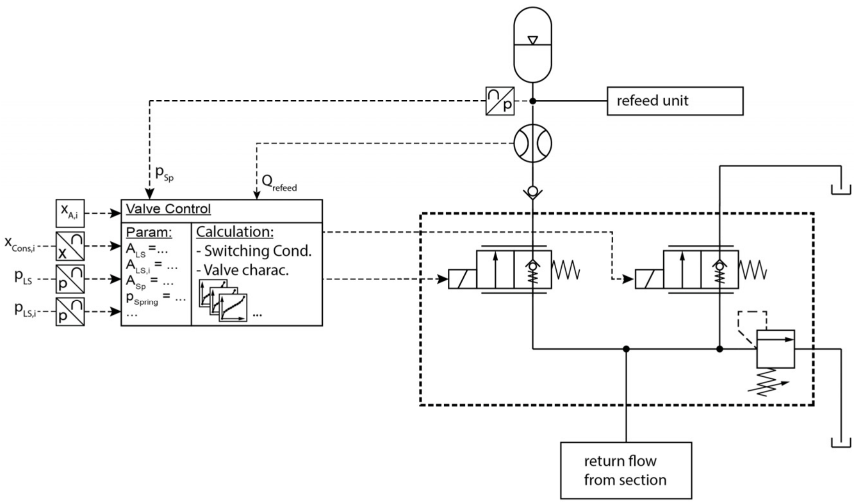


Figure 11: T/A-LV functional prototype for test rig.

In spite of the deviation of the functional prototype from the initially planned valve, the suggested design has clear advantages for test rigs. Valve control enables a simple and flexible parameterization of the functional prototype and, thus allows for parametric studies, e.g. with variable valve characteristics. To carry out such investigations using integrated hydraulic valves would only be practicable at considerable material expense and extraordinary costs. Due to the separate metering edges, the prototype moreover can be integrated easily into the test rig. All necessary parameters are already being monitored by several sensors.

The setup is highly demanding for the valve control and its ability to compensate the rather slow proportional throttle valves. Therefore, in order to determine whether the behavior of the functional prototype matches that of an integrated valve, simulations are being carried out and the results are being validated at the test rig.

8. Summary

The suggested approach can increase the efficiency of mobile machines with hydraulic LS systems, in accordance with the required boundary conditions, by reducing the system inherent pressure losses. The circuit is designed with hydraulic actuation and hence, as an advantage, does not require any additional sensors. Therefore, in order to upgrade an existing machine only few components are necessary. Each LS section must be provided with a T/A-LV and a check valve, i.e. combined in a compact module. Furthermore, each machine must be equipped with a hydraulic accumulator, a recovery unit and appropriate piping.

However, when applying the circuit to a conventional system available on the market, compatibility with the system architecture must be guaranteed. Any negative impact on the performance of the system, e.g. by switching operations, must be avoided. In addition, a cost-benefit evaluation of the efficiency improvement potential considering the machine architecture and its application scenarios or typical operating cycles should be conducted.

9. Outlook

The results obtained through simulation will be validated on the project test rig, which will be designed like an LS system of an excavator, equipped with three actuators (for further information refer to /4/). At first, synthetic simulation cycles will be used to verify the function and the performance of the circuit as well as to identify critical parameters and to optimize the circuit. Then, cycle data from /6/ will be used to validate the performance of the efficiency improved system under realistic conditions.

Problems that have not yet been solved in the present project, e.g. the design of an appropriate recovery unit and the integrated T/A-LV, shall be investigated in a following project. Subsequently, the overall system will be applied to a test vehicle and be tested in practice.

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11. Nomenclature

p_x	Pressure	1 bar
Q	Volumetric flow	l/min
x_A	Ratio of effective surfaces of a cylinder	1
F_x	Load or force	kN
A_x	Surface area e.g. at valves or in cylinders	mm²
x_i	Position	mm
v_i	Velocity	mm/s